

Reduced Order Modeling of SLS liquid hydrogen pre-valve flow guide to Enable Rapid Transient Analysis

Andrew M. Brown/ER41, Andrew Mulder /ER42

NASA/MSFC, Huntsville AL 35801

NASA is developing a new launch vehicle, called the Space Launch System (SLS), which is intended on taking humans out of low earth orbit to destinations including the moon, asteroids, and Mars. The propulsion system for the core stage of this vehicle includes four RS-25 Liquid Hydrogen/Oxygen rocket engines. These engines are upgraded versions of the Space Shuttle Main Engines (SSME); the upgrades include higher power levels and affordability enhancements. As with any new vehicle, the Main Propulsion System (MPS), which include the feedlines and ancillary hardware connecting the engines to the fuel and oxidizer tanks, had to be redesigned (figure 1 – *export clearance in progress*), as the previous MPS for the SSME's was inherently part of the Space Shuttle System, which had a completely different overall configuration

One of the most notable outstanding problematic issues from the MPS for the SSME that had to be addressed for the SLS was a significant pressure environment propagating upstream from the low pressure fuel pump called Higher Order Cavitation (HOC). HOC is a well-known cavitation-induced oscillatory phenomenon. It occurs at particular hydrodynamic conditions, i.e. flow rate, inlet pressure and blade tip speed. Higher order cavitation provides a significant oscillating load on the blades but also sends spinning pressure waves upstream at frequencies at varying, non-integer multiples of the inducer rotational speed. During the space shuttle program, these upstream-travelling waves were found to be the cause of cracking in a flowliner in the bellows of the feedline. An assessment of all the upstream structures to this excitation, therefore, was made a requirement of the SLS design.

The first step in this assessment was to characterize the HOC loading itself. This characterization was based both on an extensive measurement and analytical program to gain an understanding of the hydrodynamic field. The results were published for use by the structural design community and include a table listing discrete loadings characterized by a number of parameters. The first of these is the source, which is not only the HOC itself but also by other sources found from test, including two times HOC, HOC + 2N, and one and two times the rotational speed. The predominant frequency range of this excitation is then identified, which has approximately a bandwidth equal to 10% of the nominal frequency value, and then a description of the type of spinning wave itself, which is generally a pressure wave with 1 or 2 nodal diameters and 0 or 1 nodal circle, as specified in the equation

$$p(\theta, x, t) = a_{m,n} \cos[m(\theta - 2\pi f t)] e^{-kx}$$

This excitation is actually not a discrete sinusoid, but instead is somewhat narrow banded, so the magnitude is defined as a psi-rms value.

To assess the structural capability for the hardware impacted by this loading, the pressure wave was then applied to all the finite element models as a spinning wave at a single frequency for a number of cases corresponding to different excitations, and responses obtained. The response value was then used in fracture analysis. All the hardware in the flowpath showed acceptable stress value except for the pre-valve flow guide.

An investigation into methods for refining the analysis was therefore initiated. A number of reductions in the loading were identified, but none were large enough to obtain sufficient fracture capability. One other possible conservatism recognized was characterizing the loading as purely harmonic and running a resonant analysis at that frequency instead of using the actual narrow-band loading. The most obvious way to address this question would be to apply a transient signal directly onto the FEM. There are a number of issues with this approach, however. First of all, the transient is defined only at 4 circumferential locations, so an interpolation of these points would be required to adequately define the loading. Secondly, a transient analysis of any large finite element model is extremely computationally intensive and error-prone, and finally, the engineering manpower resources available to devote to a problem of this complexity were limited.

A unique alternative approach described in this paper was therefore developed that could be performed rapidly and would still provide a conservative but accurate assessment of the response for the loading cases. The worst case loading on the finite element model was a 2*HOC, 1 Nodal diameter wave form (called $m=1$) of amplitude 2.23 psi-rms at 3687.4 Hz, which resulted in a peak resonant response of 12474 psi (see figure 2, *export clearance in progress*). The concept for the approach was to create an equivalent single degree of freedom structure of the FEM, calibrate loading on the SDOF to the FEM response using the harmonic analysis, and then apply the true loading using a Matlab script to quickly obtain a stress time history.

The first step in this process is to obtain the actual pressure time histories that were used to generate the 2.23 psi-rms forcing function. These time histories, which were measured at a plane close to the inducer (only 0.4 length/diameter units upstream) for four different circumferential locations for a 60 second period, were filtered with a passband of 3300-3700 Hz. In addition, to further reduce excess conservatism, since the 1st nodal diameter waves are the only type that travel all the way upstream to the pre-valve, and there were four locations, Nyquist criteria allow these histories to be spatially decomposed with a 2D Fourier Transform into a single complex time history containing the nodal diameter 1 content, plotted in figure 3.

Assuming a SDOF system with unit mass and natural frequency equal to a value iterated to obtain maximum response (further discussion below), the equivalent force F_{eq} necessary to excite the SDOF such that the response equals the rms stress response σ_{FEmax} from the FEM is obtained using

$$X_{peak} = \frac{F_{eq}/k}{2\zeta} = \frac{F_{eq}/m\omega^2}{2\zeta} = \frac{F_{eq}}{2\zeta\omega^2}$$

$$\sigma = (\text{Constant})X_{peak},$$

So, for $\zeta = .006$, and a natural frequency of 3556 Hz (in this particular example, this is the Fourier peak of a particular time history),

$$F_{eq} = (\text{Con})\sigma_{FE\max} 2\zeta\omega^2 = (\text{Con})(12474)(2)(.006)(2*\pi*3556)^2 = 7.4726e10$$

The constant shown here is used to indicate the the actual response sought is a stress, but that the displacement response obtained with this equation is linearly related to stress, so the constant can be ignored in this procedure. The Matlab script ODE45u to be used for this procedure was verified by generating a harmonic excitation of this magnitude, applying it to the SDOF and obtaining the response, which matched the sought-for response of 12474 psi, as expected.

Since F_{eq} , the force applied to the SDOF, is equivalent to the harmonic force applied to the FEM, the ratio of F_{eq} to that original harmonic forcing function is then calculated for use as a calibration constant.

$$ratio = \frac{F_{eq}}{Amp_{sin}} = \frac{7.4726e10}{2.23} = 3.3509e10$$

The spatially decomposed $m=1$ pressure time history $P_{m=1}(t)$ is complex, so the real and imaginary parts can be resolved to yield the amplitude for any circumferential location.

$$p(t, \theta) = \text{Re}(P_{m=1}(t)) * \cos(\theta) + \text{Im}(P_{m=1}(t)) * \sin(\theta)$$

The finite element model to SDOF simplification requires that the 3-D loading be lumped into a single point application, in this case the circumferential location. Since it is unknown a-priori which location will yield the worst stresses, a loop was written to calculate transients for 10 degree increments. To apply this load, these transients are multiplied by the calibration ratio, applied to the SDOF using the Matlab script, and a stress response generated. In addition, a worst-case response was initially thought to be when the SDOF has a natural frequency equal to the peak frequency of the transient as identified by a Fourier Transform of the transient. Several iterations showed that the peak was not always at this value though, because of non-stationary of the data and the slight theoretical variation of peak response from natural frequency. Another nested loop was therefore written to obtain the response time history for a range of possible natural frequencies.

To determine the worst of all the circumferential and natural frequency cases, the maximum value was obtained from the time history and plotted in Figure 4. The result is a peak response at 180 degrees

at a natural frequency of 3543.5 Hz. The individual transient excitation, scaling, and stress time history response for that worst case are shown graphically in Figure 5.

The benefit of performing this analysis, though, is in the application of the variation of stresses over the life of the component rather than assuming it responds at a single peak stress. This is accomplished by producing a histogram of the response stress for the above location as shown in Figure 5. This histogram was converted to a PDF and used to generate the stress life history for the operational life for points in the FEM along possible fracture paths. In addition, since the HOC is not at full strength during the entire operational life, this stress distribution was scaled further by the strength levels of the HOC which occurred at different durations. With the application of the stress life history from the SDOF analysis and this final convolution with the HOC strength, the final fracture analysis shows the part has acceptable life and can be fabricated as designed. The application of this unique, tractable, reduced order methodology has enabled the SLS program to avoid substantial cost and schedule penalties if a redesign or change of material were required, as had been considered.

Note: The final SciTech paper will provide more theoretical justification of the SDOF method, will examine potential error in the analysis, and will provide suggestions for (or results of) further verification analysis. In addition, clearance for some more illustrative figures will hopefully be obtained for the paper itself.

Figure 1: Fuel Feedline Configuration

Figure 2. Finite Element Model of Pre-Valve

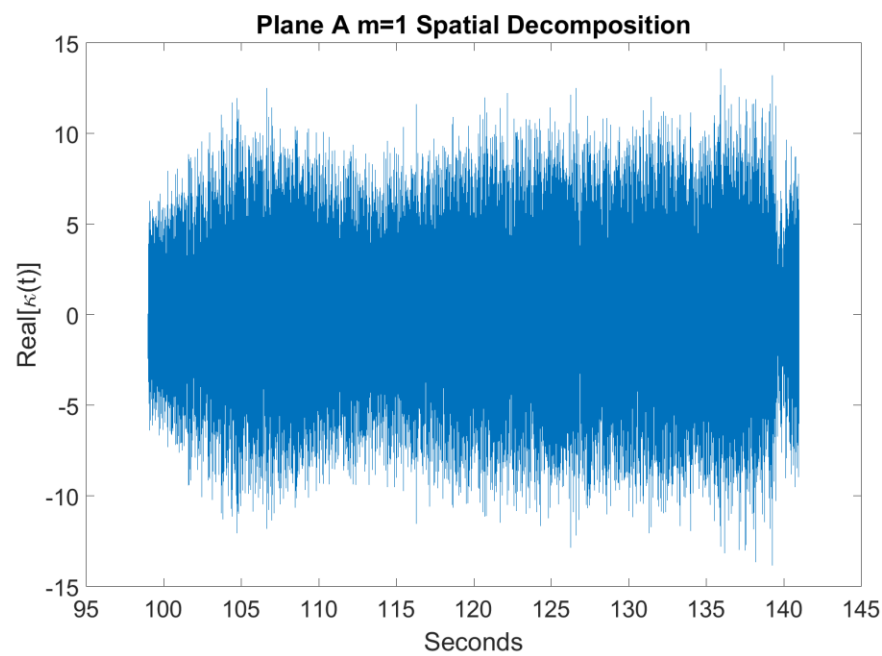


Figure 3. 1st Nodal Diameter Spatial Decomposition of measured pressures at plane A

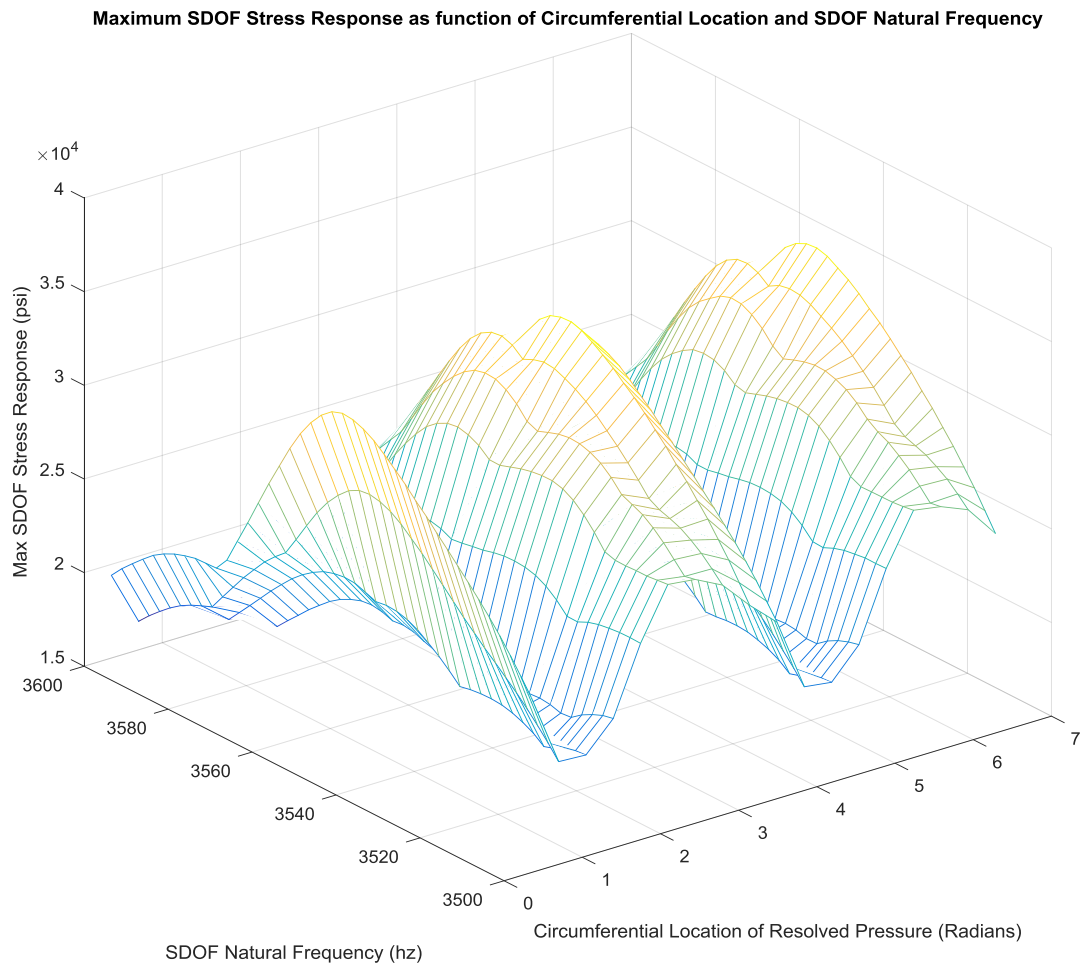
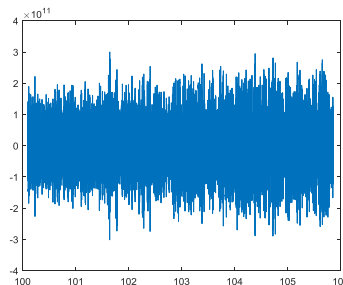


Figure 4. Maximum Transient Stress Response of SDOF as function of Circumferential Location and SDOF Natural Frequency

Measured
Transient
Data

$\times 3.3509e10 =$



Matlab script
ODE45u onto
SDOF

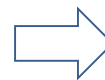
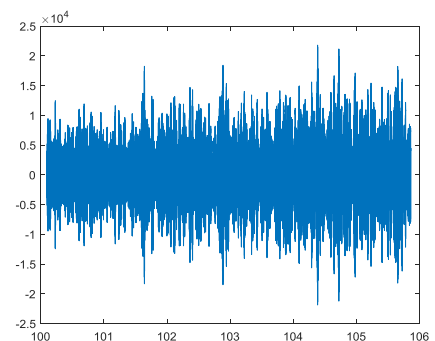


Figure 5 – Transient Excitation multiplied by Scaling Factor,
Resulting in Stress Time History



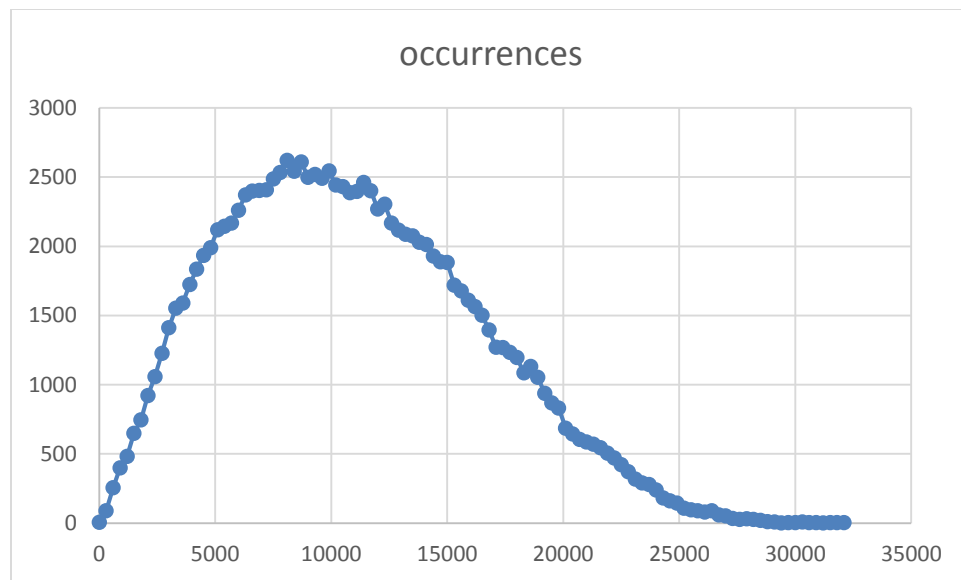


Figure 6. Histogram of Transient Stress for Peak SDOF